SCHUSTER

Design of a pneumatic mandrel press

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DESIGN OF A PNEUMATIC MANDREL PRESS

BY

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THIS IS TO CERTIFY THAT THE THESIS PREPARED UNDER MY SUPERVISION BY

George Schuster

ENTITLED Design of a Pneumatic Mandrel Press

IS APPROVED BY ME AS FULFILLING THIS PART OF THE REQUIREMENTS FOR THE

DEGREE OF Bachelor of Science in Mechanical Engineering

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DESIGN OF A PNEUMATIC MANDREL PRESS.

Specifications.

To design a pneumatic mandrel press fulfilling the follow-
ing specifications.

Depth of Throat	13 inches.
Stroke of Ram	8 "
Distance from Bearing Plate to the Ram when the Ram is at the Top of its Stroke . .	16 "
Air Pressure	100 pounds.
Capacity of Press	5 tons.
Maximum Size of Mandrel	3 inches.

In order to provide for mandrels of various lengths, an ad-
justment is to be provided making it possible to lengthen the
reach of the ram at least two inches.

The ram shall be returned to the top position by means of
air pressure, and a means is to be provided whereby the air supply
is automatically shut off when the ram is in either top or bottom
position.

The frame of the press shall be made of cast iron, having
a T-section, and reinforced as shown on drawing No.II.

The air cylinder shall be made of close grained cast iron.

The stresses for cast iron shall not exceed the following:-

|	
Max. Tensile Stress . . .	2,000 lbs. per sq. inch.
Max. Compression Stress . .	4,000 " " "
Max. Shear	2,000 " " "

Fig. 1.
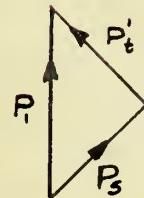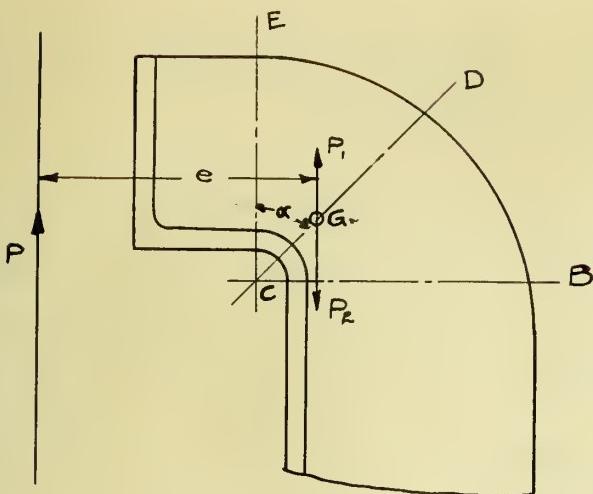
Force Analysis.

    In order to determine the proper size of the frame at different sections, it is necessary to determine the forces and stresses acting on the sections considered. Referring to Fig. I, the section at CD, making the angle α with the line of action of the ram, will be the first one considered.

    Let P = the force acting along the center line of the ram.

        e = the distance in inches from the center line of the ram to the gravity axis of the section. (Projected as a point at G).

        I = the moment of inertia of the section with respect to the gravity axis of that section.

        c_t = the distance of the remotest fiber, in tension, from the neutral axis.

        c_c = the distance of the remotest fiber, in compression, from the neutral axis.

    Then the tensile stress is given by the expression

$$S_t = \frac{Pec_t}{I} \quad . \quad . \quad . \quad . \quad . \quad (1)$$

2

and the compressive stress by

$$S_c = \frac{Pec_c}{I} \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

Let the forces, P_1 and P_2 each equal to P, act through G, the projection of the gravity axis CD as shown. Then the force P_1 may be resolved into two components, one acting along the section and the other at right angles to it. The force triangle shows these components. The shearing stress acting along the section is given by the expression

$$S_s = \frac{P\cos\alpha}{A} \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

where A equals the area of the section.

The uniform tension over the section due to the component at right angles to CD is given by

$$S'_t = \frac{P\sin\alpha}{A} \quad . \quad . \quad . \quad . \quad . \quad (4$$

The equations as given apply for the angular section CD. For a vertical section through C, as along CE, α becomes zero and the tensile stress reduces to zero. The shearing stress becomes

$$S_s = \frac{P}{A} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

For a horizontal section through C, as along CB, the angle α becomes 90 degrees. For this case the shearing stress reduces to zero and the uniform tension is given by

$$S'_t = \frac{P}{A} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

The following formulas from Merriman's "Mechanics of Materials" represent a combination of the expressions given above and are used for determining the maximum tensile, compressive, and shearing stresses for the sections to be considered.

Max. Tension $= \dfrac{S_t + S'_t}{2} + \sqrt{S_s^2 + 1/4(S_t + S'_t)^2}$ . . (7)

3

Max. Compression $= \dfrac{S_c - S'_t}{2} + \sqrt{S_s^2 + 1/4(S_c - S'_t)^2}$ . . (8)

Max. Shear $= \sqrt{S_s^2 + 1/4(S_t + S'_t)^2}$ or $\sqrt{S_s^2 + 1/4(S_c - S_t)^2}$ (9)

In the calculations for the sections along CE,CD,and CB presented on the following pages,the tables for each section give the properties of that section,and the values are substituted in the foregoing formulas directly without reproducing the formulas as derived above.

The values of e,the distance from the line of action acting through the ram to the gravity axis of the section,are shown for each section. The force acting is taken as five tons,or 10,000 pounds. Fig.2 shows the general arrangement of the factors considered.
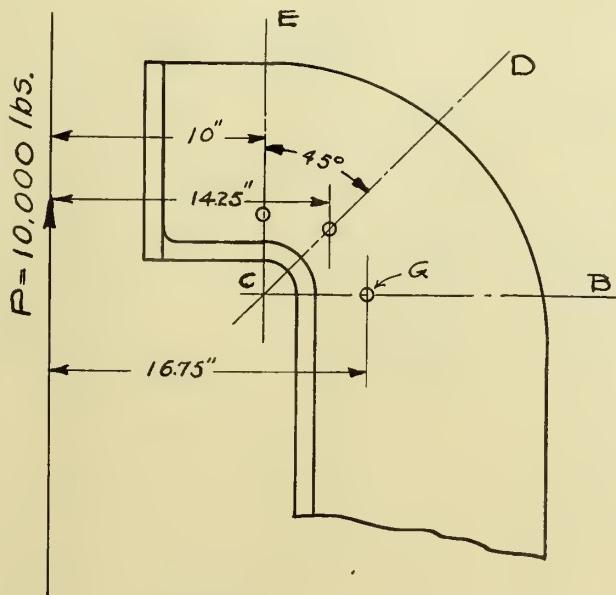

FIG. 2.

CALCULATIONS.

Sections of Frame.

Horizontal section.
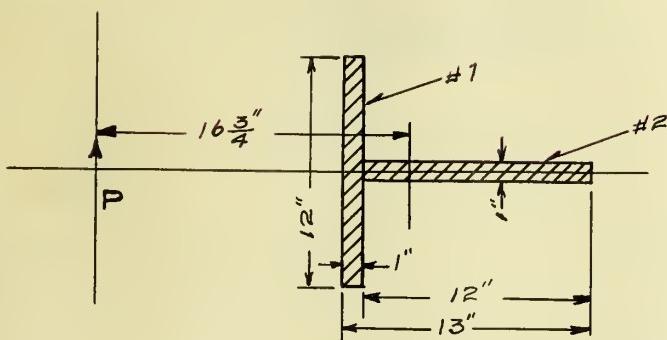
No.	Area Sq.in.	Mom. Arm in.	Mom.	c_t	c_c	I_{cg}	h in.	Ah^2	I	$\frac{I}{c}$	
										Ten.	Comp.
1	12	.5	6	3.75	9.25	1	3.25	127	128	106	431
2	12	7	84			144	3.25	127	271		
	24								399		

Substituting the proper values in equations (1),(2),(6),(7), and (8), we get the following results

Tensile stress, S_t = 1,580 lbs.per sq.inch.

Compressive stress, S_c = 3,880   "   "   "

Uniform Tension, S_t' = 420   "   "   "

Maximum tension = 2,000   "   "   "

Maximum compression = 3,460   "   "   "

Angular(45 degree) Section.
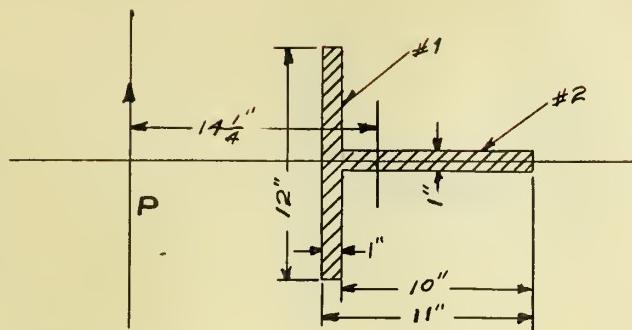
No.	Area Sq.in.	Mom. Arm in.	Nom.	c_t	c_c	I_{cg}	h in.	Ah^2	I	$\dfrac{I}{c}$ Ten.	$\dfrac{I}{c}$ Comp.
1	12	.5	6			1	2.5	75.	76	106.4	39.91
2	10	6	60	3	8	83.3	4	160	243.3		
	22		66						919.3		

Substituting the proper values in equations (1),(2),(3),(4), (7),(8),and(9),we get the following results

Tensile stress,S_t = 1,340 lbs. per sq. inch.

Compressive stress,S_c = 3,580 " " "

Shearing stress,S_s = 322 " " "

Uniform tension,S_t' = 322 " " "

Maximum tension = 1,740 " " "

Maximum compression = 3,280 " " "

Maximum shear = 1,650 " " "

Vertical Section.

No.	Area Sq.in.	Mom. Arm in.	m.	c_t	c_c	I_{cg}	h in.	Ah^2	I	$\frac{I}{c}$ Ten.	Comp.
1	12	.5	6			1	2.14	55	56		
2	9	5.5	49.5	2.64	7.36	60.6	2.86	73.8	133.8	71.8	25.7
	21		55.5						189.8		

Substituting the proper values in equations (1),(2),(5),(7), (8),and (9), we get the following results

Tensile stress,S_t = 1,390 lbs. per sq.inch.

Compressive stress,S_c = 3,900 "   "   "

Shearing stress,S_s = 476 "   "   "

Maximum tension   = 1,535 "   "   "

Maximum compression = 3,950 "   "   "

Maximum shear   = 2,000 "   "   "

Cylinder Calculations.

From the specifications,we have the following data:-

P = 10,000 lbs., Air pressure = 100 lbs. per sq.inch.

Allowing 10% for friction,the total pressure is 11,000 lbs, hence the area of the cylinder in square inches will be

$$A = \frac{11,000}{100} = 110$$

from which the diameter of the cylinder in inches will be approxi-
mately 12.

The thickness of the cylinder walls as given by Whitham's
formula is

$$t = .03\sqrt{PD} \quad . \quad . \quad . \quad . \quad . \quad . \quad (10)$$

where P = the pressure in pounds per square inch.

D = diameter of cylinder in inches.

t = thickness of walls in inches.

Substituting the known quantities in (10),we find that

$$t = .03\sqrt{100\cdot 12} = 1.03 \text{ inches.}$$

In order to allow for counter boring and permit reboring,
this thickness will be increased to 1 1/4 inches.

The thickness of the cylinder heads as given by Merriman's
formula is

$$d = r\sqrt{\frac{3R}{4T}} \quad . \quad . \quad . \quad . \quad . \quad . \quad (11)$$

where r = inside radius of the cylinder in inches.

R = air pressure in pounds per square inch.

T = working stress of material.

d = thickness of head in inches.

Substituting the known quantities in (11), we have

$$d = 6\sqrt{\frac{3\cdot 100}{4\cdot 2000}} = 1.18 (\text{say } 1\ 1/8 \text{ inches}).$$

Cylinder Head Screws.

In calculating for strength,the screws will be considered
in tension. The total force equals

$$.785\cdot 144\cdot 100 = 11,300 \text{ pounds.}$$

The force on one screw,assuming six screws are used,is

$$11{,}300/6 = 1{,}880 \text{ pounds,}$$

from which it follows that the area of the screw must equal

$$\frac{P}{S} = \frac{1{,}880}{5{,}000} = .375 \text{ square inches.}$$

This area corresponds to that of a screw 3/4 inch in diam.

Cap Screws for holding Cylinder to Frame.

In calculating the size of these screws, they will be considered as subjected to a shearing action. The force causing the shear is 10,000 pounds. Hence the shearing force per screw, assuming six screws are used, is 1,667 pounds from which we get its area as .333 square inches. This area corresponds to that of a screw 3/4 inches in diameter.

In the above calculations, the working stress, S, is taken as 5,000 pounds per square inch. This value is rather low but is in accordance with accepted practice.

Ram and Adjusting Screw Calculations.

Since the adjusting screw is short, it will be treated as a member in compression.

Let P = the force on the ram.

S = the unit working stress.

A = adjusting screw root area.

Then $A = \dfrac{P}{S} = \dfrac{10{,}000}{8{,}000} = 1 \ 1/4$ square inches, and the corresponding diameter at the root of the thread is 1 1/4 inches.

Adjusting Rod Thread.

A square thread will be used of 1/4" pitch, or four threads per inch. The area at the root equals

$$1/8 \cdot 3.92 \cdot 4 = 1.96 \text{ square inch of surface}$$

per inch of thread.

The least area to prevent stripping will be

$$A = P/S = \frac{10,000}{8,000} = 1 \ 1/4 \text{ square inches.}$$

So at least one inch of thread on the screw must be in contact with the cast iron thread in the ram to prevent stripping. The outside diameter of the screw will be 1 1/2 inches.

Cast Iron Thread in the Ram.

The area at the root of the thread will be

$$1/8 \cdot 4 \cdot 4.71 = 2.36 \text{ square inches of surface}$$

per inch of thread.

The least area to prevent stripping will be

$$A = P/S = \frac{10,000}{2,000} = 5 \text{ square inches.}$$

Then the length of the cast iron thread in contact with the screw, measured axially is 2 1/4 inches.

Hence the adjusting screw must be at least 2 1/4 inches long inside of the ram, when the rod is in its lowest position. For a two inch adjustment, it follows that the total length of screw inside of the ram must be five inches. This will be exclusive of the length of the screw for extending through stuffing box and adjusting hand-wheel.

Ram.

Since the outside diameter of the screw is 1 1/2 inches, 2 1/2 inches will be assumed as the outside diameter of the ram. This will give a diameter which will not require an excessively

large stuffing box,and at the same time keep down the weight   of
the ram.

Since the piston rings(2 in number) are each 1/2 inch wide,
and allowing for proper spacing with sufficient material between
the edge of the groove and the edge of the piston,the proper thick-
ness of the piston will be three inches.

20"

3'-4"

13"

9'-2"R

Section at A-B

Section at E-F

Section at C.D.

Main Frame
No. 1

1" Core

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MANDREL PRESS
SCALE 1½:4
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THESIS
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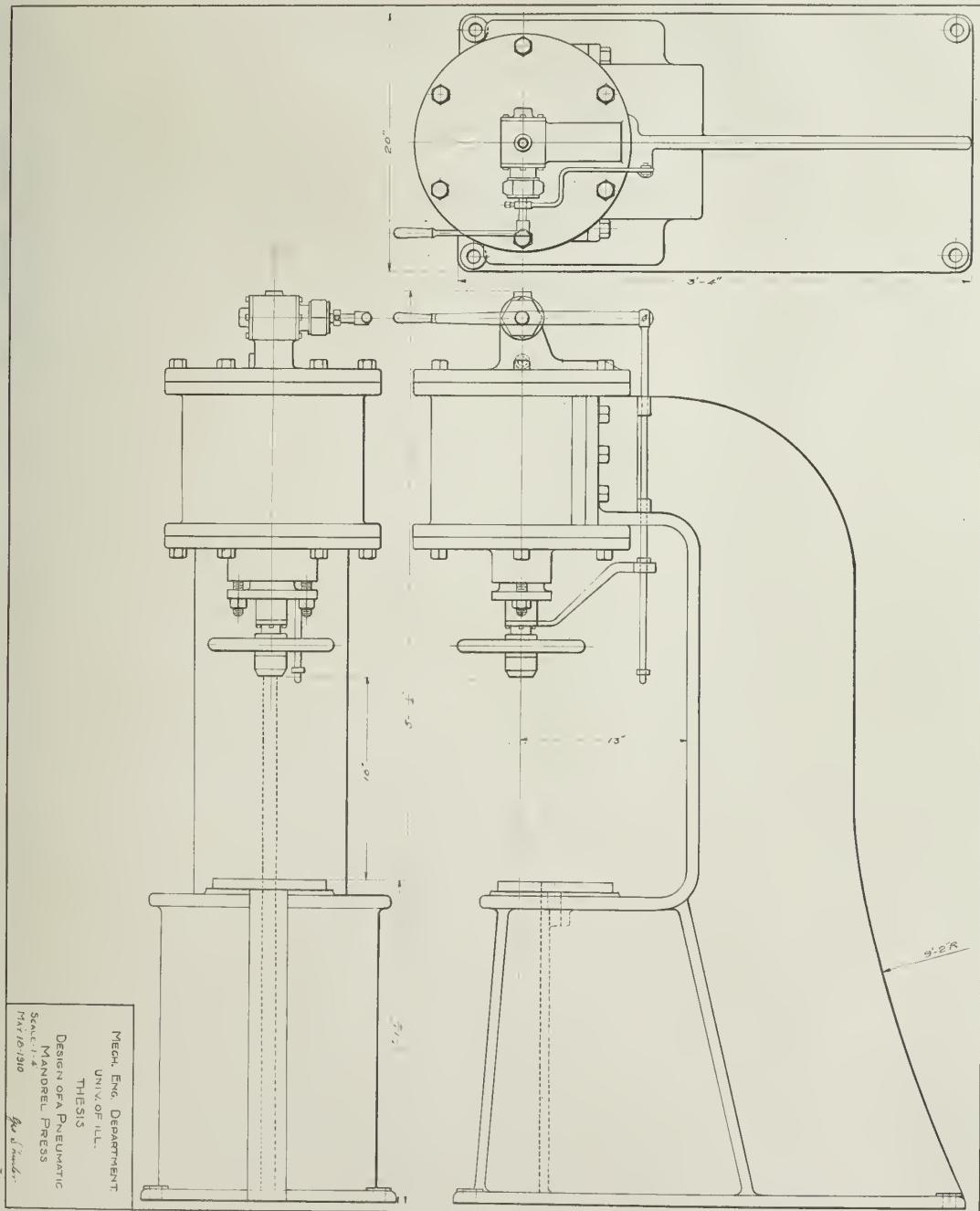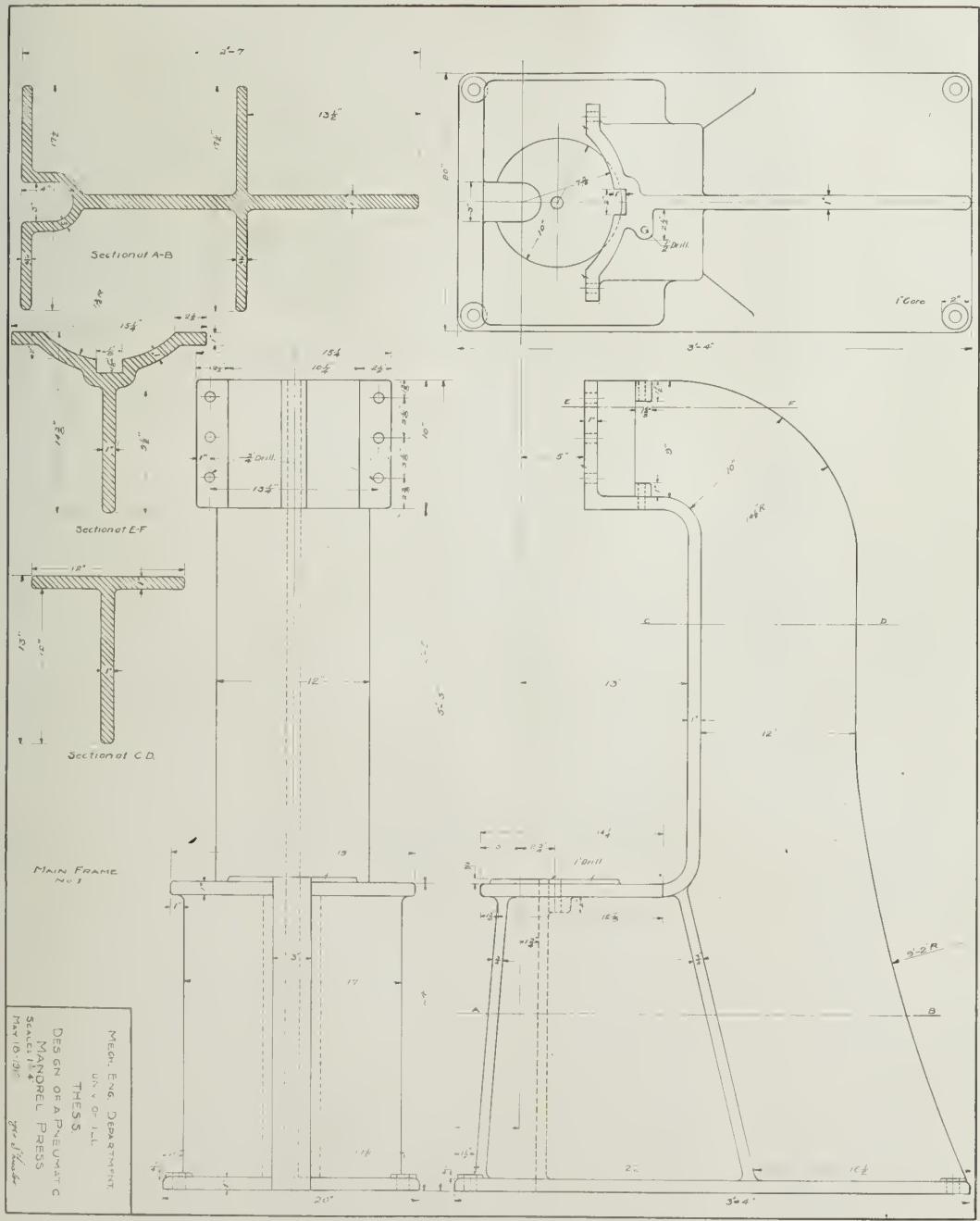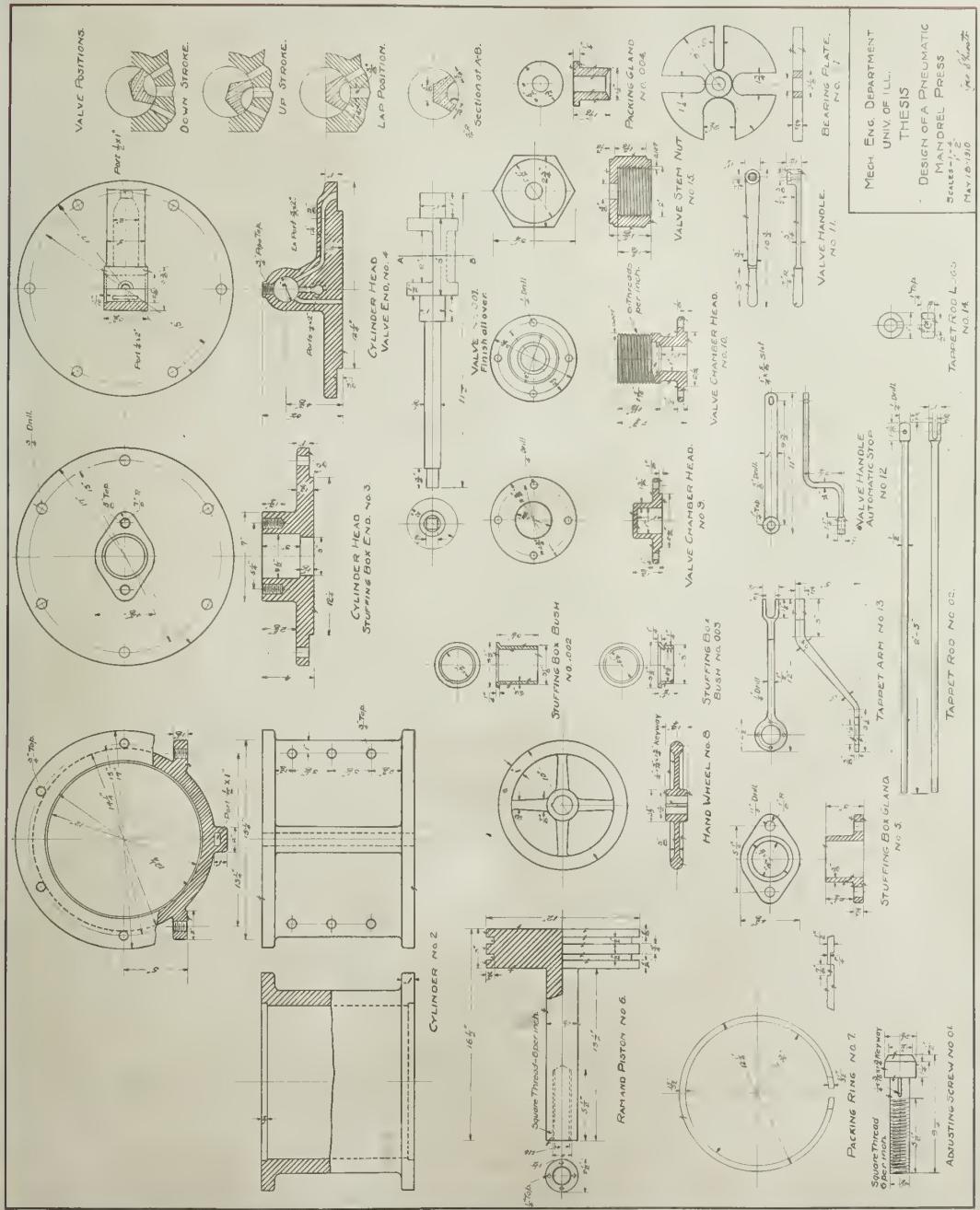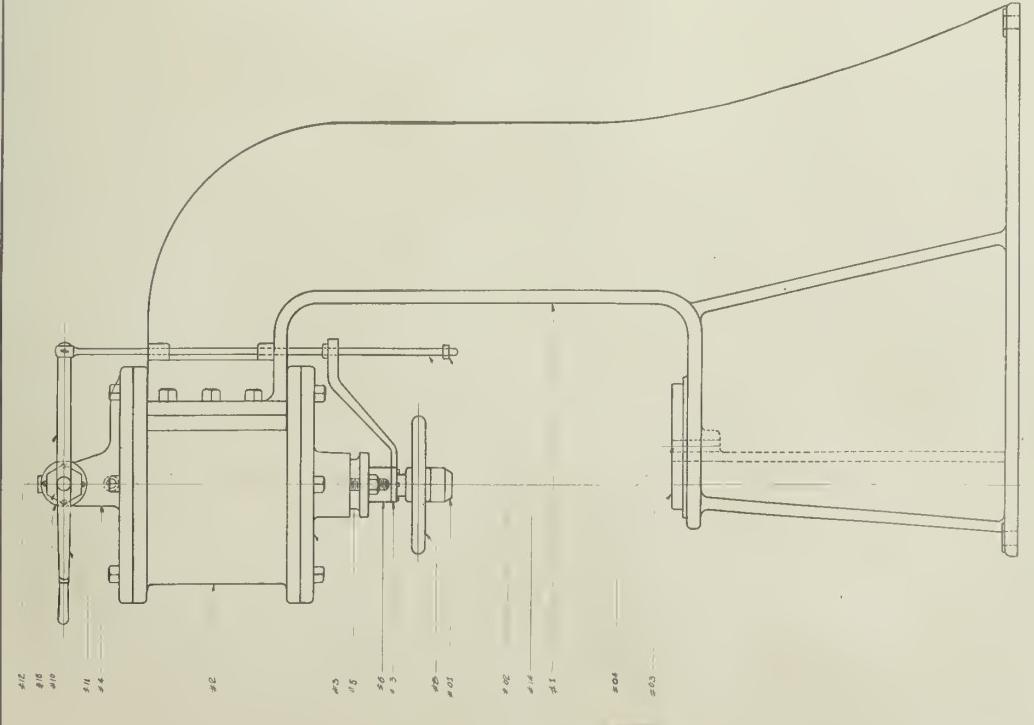
BILL OF MATERIALS AND KEY TO ASSEMBLY

No	Name	Sheet No.	Number Wanted	Remarks
CAST IRON PARTS				
1	Frame	II	1	
2	Cylinder	III	1	
3	Cylinder Head	III	1	Ram End
4	Cylinder Head	III	1	Valve End
5	Stuffing Gland	III	1	For Head No.3
6	Ram and Piston	III	1	
7	Packing Ring	III	2	For Piston
8	Hand Wheel	III	1	Adjusting Screw
9	Valve Chamber Head	III	1	
10	Valve Chamber Head	III	1	Packing Nut End
11	Valve Handle	III	1	
12	Stop Lever	III	1	
13	Tappet Arm	III	1	
14	Tappet Rod Lugs	III	2	
15	Valve Stem Nut	III	1	
BRASS PARTS				
001	Valve	III	1	
002	Stuffing Box Bush	III	1	Stuffing Gland
003	Stuffing Box Bush	III	1	For Head No.3
004	Packing Gland	III	1	Valve Stem
STEEL PARTS				
01	Adjusting Screw	III	1	
02	Tappet Rod	III	1	
03	Bearing Plate	III	1	Automatic Stop
04	Bearing Plate Pin	III	1	

STANDARD PARTS

Name	Size	Number Wanted
Cap Screws	¾" × 2¼"	6
Cap Screws	¾" × 2"	12
Mach Screws	⅜" × ⅝"	8
Mach Screws	¼" × ⅝" × ⅝"	4
Studs	⅜" × 5"	2
Nuts		2
Set Screws	¼" × ¾"	3
Pin		1
Cotter Pins	3½" × 2"	2